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Rollover Protective Structure (ROPS) Performance Criteria for Large Mobile Mining Equipment

By Stephen A. Swan

UNITED STATES DEPARTMENT OF THE INTERIOR

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CONTENTS

ILLUSTRATIONS

TABLES

UNIT OF MEASURE ABBREVIATIONS USED IN THIS REPORT

°/s	degree per second	lb	pound
ft	foot	pct	percent
g	acceleration of gravity	psi	pound per square inch
in	inch	s	second
in•lb	inch-pound		

ROLLOVER PROTECTIVE STRUCTURE (ROPS) PERFORMANCE CRITERIA FOR LARGE MOBILE MINING EQUIPMENT

By Stephen A. Swan [1]

ABSTRACT

Certain types of mobile surface mining equipment are required to be equipped with rollover protective structures (ROPS) capable of protecting the operator in case of a rollover accident. The Bureau of Mines has developed ROPS performance criteria for wheeled front-end loaders. This report reviews the development of rollover test procedures and subsequent testing of ROPS for 52,000-, 286,000-, and 390,000-lb gross vehicle weight (GVW) loaders. The program consisted of both dynamic (roll) and static tests to assist in the development of the ROPS performance criteria. Each test included extensive instrumentation and photographic documentation to fully evaluate the performance of the ROPS. Data summarizing side, longitudinal, and vertical force and deflections are included. The data indicate that for loaders in excess of 240,000 lb, GVW, a side force-to-mass ratio of 1.0 and an energy-to-mass ratio of 6.0 appear to be adequate criteria to protect an operator in the event of a 540° rollover. For smaller machines, Society of Automotive Engineers (SAE) J1040 performance criteria provide adequate protection. Longitudinal loading, though not addressed in previous ROPS performance criteria, was shown to be a potentially significant factor in ROPS failure. Longitudinal loading and energy criteria equal to 80 pct of side loading criteria are recommended.

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INTRODUCTION

Mine Safety and Health Administration (MSHA) and Occupational Safety and Health Administration (OSHA) regulations require that certain types of mining, construction, earthmoving, agricultural, and forestry equipment be equipped with a rollover protective structure (ROPS). The types of mobile machines that must be equipped with ROPS include crawler tractors and crawler loaders, motor graders, wheeled loaders and tractors, skid-steer loaders, and the tractor portion of tractor-scrapers. Although not mandated by regulation, ROPS are sometimes installed on off-highway haulage trucks and water trucks.

The requirement of ROPS regulations in the United States, Canada, and other countries is due to the significant injury potential represented by accidental rollovers of mobile equipment during field use. The use of ROPS and seatbelts on these types of equipment can greatly reduce the number and severity of injuries resulting from rollover accidents. Although the regulations require that the employer who owns the machine equip it with ROPS to provide a safer work environment for employees, manufacturers of the machines typically install ROPS on the machines before ownership is transferred to the employer.

The Society of Automotive Engineers (SAE), through Subcommittee 12—Off Road Machinery Technical Committee (ORMTC), develops ROPS structural performance and test method criteria for use by industry in the design and performance certification of ROPS used on construction and mining machines. The ROPS regulations promulgated by OSHA and other regulatory bodies base ROPS structural performance capability on criteria developed by SAE.

In 1972, ORMTC began development of SAE Recommended Practice J1040 "Performance Criteria for Roll Over Protective Structures (ROPS) for Construction, Earthmoving, Forestry, and Mining Machines." However, because of the absence of rollover test data, the force-energy criteria for machines exceeding 132,000 lb gross vehicle weight (GVW) were projected based on the collective judgment of the committee. The committee reasoned that ROPS force-energy criteria, expressed as a function of total machine size and weight, could be less for large machines than for small machines. The lower requirement was possible because the operator compartment on a larger machine is smaller, in proportion to total machine size, than for a small machine. As a result, when a large machine rolls, a greater proportion of the energy generated by the roll would be absorbed by the machine body, tires, frame, etc., than would be the case for a small machine.

In 1977, rollover test data were produced indicating that the force and energy requirements contained in the SAE J1040 recommended practice were not sufficient for large machines. A machine manufacturer performed ROPS rollover tests on a front-end loader and crawler tractor, each weighing approximately 200,000 lb. Although both ROPS exceeded SAE J1040 performance criteria, neither survived the roll test. SAE subsequently informed OSHA that the validity of the force and energy requirements delineated in J1040 was in doubt and recommended that new guidelines be developed for larger construction and mining machines. An interim guideline, SAE J1040c, was established by ORMTC, however it, like its predecessor, was based on subjective judgments and extrapolation from available test data. As larger and heavier front-end loaders were introduced (up to 390,000 lb), the adequacy of these judgments and extrapolations was seriously questioned. This report documents research to establish ROPS performance criteria for large front-end wheeled loaders through a systematic program of dynamic (roll) and static testing. The test program involved trials with loaders weighing 52,000, 286,000, and 390,000 lb. The research was performed by Woodward Associates, Inc., under contract to the Bureau.[2]

TEST PROCEDURES

Because the objective of the program was the establishment of ROPS performance criteria for large front-end wheeled loaders without reliance on subjective judgments and extrapolation from tests of smaller machines, emphasis was placed on full-scale testing of ROPS under realistic, yet controlled and reproducible conditions. Roll testing, wherein a host machine equipped with a fully instrumented ROPS is caused to roll down an embankment, is the most realistic test possible, as it nearly duplicates field conditions. However, not all test variables can be easily controlled. Thus, "identical" rolls may produce a range of data values that require considerable interpretation to provide meaningful results. Static testing, wherein the ROPS is installed in a load frame that applies measured loads of known magnitude and direction, produces results which are more reproducible and there is less uncertainty in the data. However, the magnitude and direction of the loads that must be applied in order to predict ROPS performance under field conditions are not always known. Roll testing is also significantly more costly than static testing.

As a result of these considerations, a test plan was devised that incorporated both roll testing and static testing. ROPS performance criteria would be based on the combined results of the two procedures. Depending on the degree of correlation between results of the roll and static testing, it was also hoped that greater reliance could be placed on static testing in the future, perhaps in conjunction with computer simulation, as a way of reducing the cost of assessing the performance capability of ROPS structures.

The following sections describe the characteristics of the test fixtures and equipment and the procedures utilized during the test program.

ROLL HILL CHARACTERISTICS

Roll Hill Slope and Vertical Drop Height

Two considerations govern selection of roll hill slope: side loading of the ROPS at initial impact and continuation of the roll to achieve at least 360° machine rotation. The critical design load for a ROPS is generally in the side direction because of high bending stresses in the support members. Thus, shallower slopes that maximize side loading of the ROPS at initial impact are favored by some equipment manufacturers. SAE J1040c stipulates a 30° maximum slope to

[2] Dahle, J. L., and G. R. Gavan. Development of Rollover Protective Structures (ROPS) Performance Criteria for Large Mobile Mining Equipment (contract H0292020, Woodward Associates, Inc.). BuMines OFR 57-86, 1985, 277 pp.; NTIS PB 86-216066.

insure high initial side loads. However, if the slope is too shallow, the machine simply tips over and slides down the slope without rolling. The manufacturers with the most roll testing experience favored 30° to 40° slopes because they found the machines would reliably roll only on steeper slopes. It was concluded that a compromise roll hill slope of 35° would produce acceptable initial side loading while insuring at least 360° roll rotation.

Another test parameter related to roll hill slope is the vertical drop height from the launch pad to the slope. Low vertical drop heights, like shallow roll hill slopes, introduce high initial side loads, which is desirable. However, higher drop heights induce greater angular momentum in the machine, which is necessary to cause rotation and successful rolling. Analysis indicated a 30-in vertical drop height would produce the desired results.

Roll Hill Length

There was general agreement among the manufacturers consulted that the length of the roll hill should be based on two factors—the perimeter of the machine to be rolled and the number of expected rolls. It was recommended that an additional 30 pct of hill length would be required to account for any machine slippage during the roll. Therefore, it was determined that a length of 120 ft would accommodate a minimum of 720° of roll from the smallest to the largest machine to be tested (fig. 1).

ROLL-TEST PROCEDURES

All manufacturers consulted agreed that the machine should be positioned at the top of the slope with the outermost wheels at the edge of the slope. The machine would then be tipped to the side to initiate roll down the hill. This method, as opposed to the machine having a forward motion when tipped over the edge of the hill, would provide for the best repeatability of test conditions.

Method of Release

The machine was positioned at the top edge of the slope on a tilt table. The side of the platform away outboard from the edge of the roll hill was elevated (fig. 2) and the machine rolled off the platform when the center of gravity passed beyond the inboard tires. The advantage of using a tilt table, in addition to providing repeatable release conditions, was that it could be reused for other testing.

Launch Pad

A launch pad for the machines was necessary in order to provide the method of release as described above. The inboard side of the tilt table had to be pinned to the structure in order to rotate. In addition, as the machine is being tipped all of its

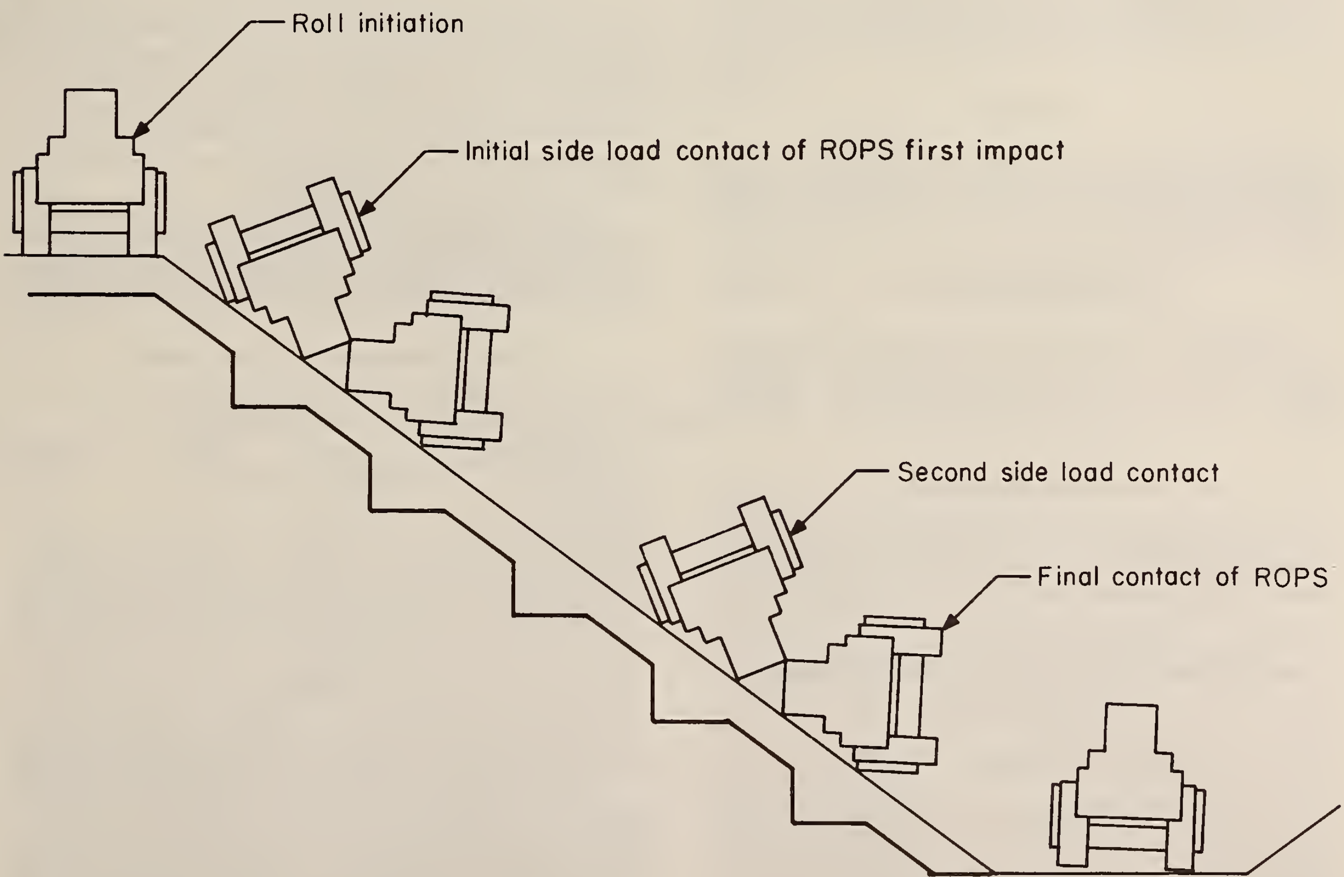

Figure 1. — Equivalent rollover, 720°.

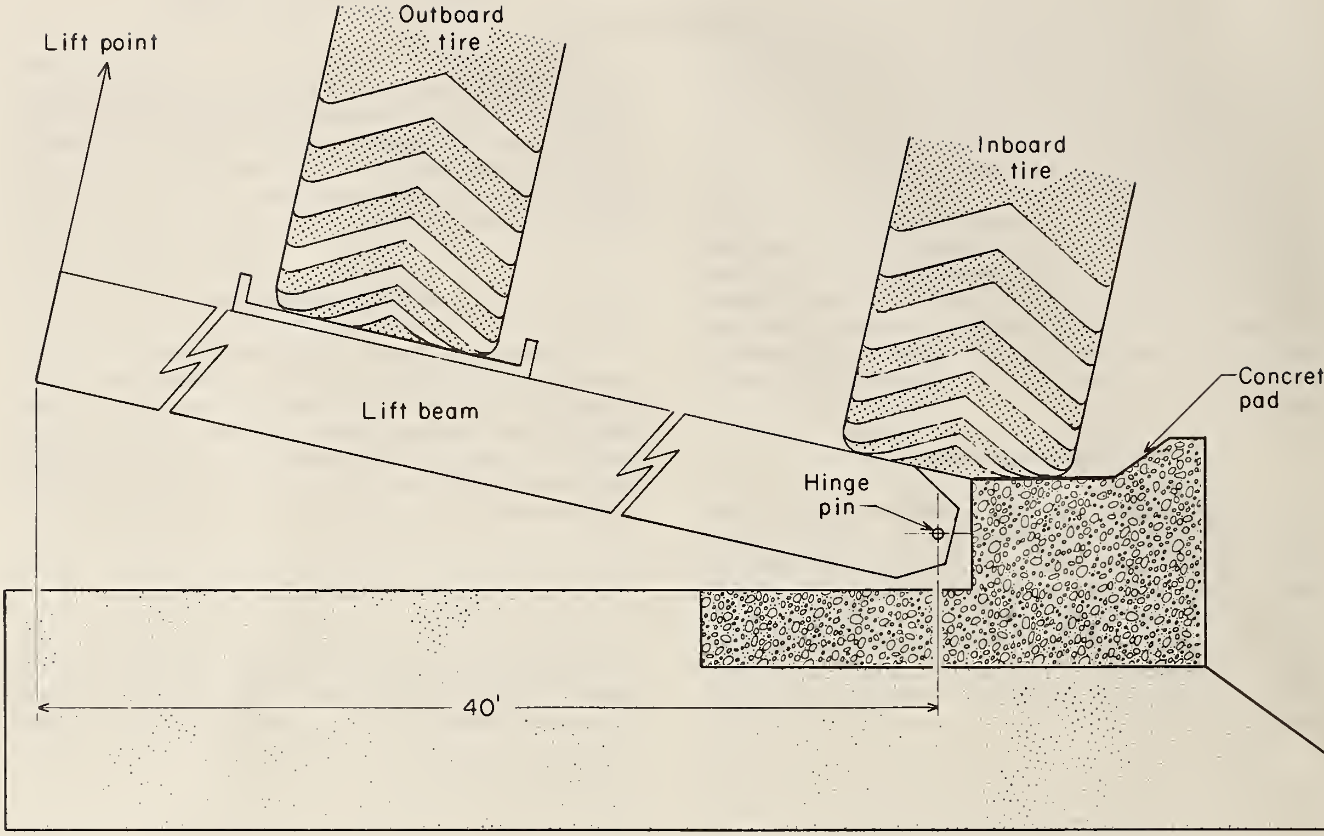

Figure 2. — Tilt table.

weight is on the two inboard tires. A launch pad or footing was necessary to prevent the edge of the hill from slipping or falling away (fig. 2).

Bucket Position

During all tests the bucket was locked and welded in the carry position as defined by SAE J732c. This definition requires the bucket to be tipped backward to a 15° approach angle.

Articulation Joint

The articulation on all three machines was locked by the transportation locking bars and additional supports were welded in place. Again this was to provide for a repeatable test.

INSTRUMENTATION AND MEASUREMENT PARAMETERS

Data Link and Sensor Types

The type of data link was an important factor in establishing the other components of the instrumentation system. It was necessary to define the data link early in the planning of the instrumentation system (fig. 3). The types considered were the umbilical (hard-line) type or a telemetry system with AM-FM transmission. The umbilical system was selected as the best data link. The recording system and components were based on the use of this method.

Once the measurement parameters were determined, as described in the following sections, it was possible to select the type of sensors necessary to meet the program objectives. The sensors selected to measure the parameters are listed in table 1.

Table 1.—Parameter measuring sensors

Instrumentation	Umbilical Front	Umbilical Rear	Channels
Accelerometer	3	3	6
Potentiometer	3	3	6
Rate gyroscope	1	1	2
Strain gauge	10	10	20
Time generator	1	1	2

Strain

Strain was considered to be the most important parameter. Of the 36 channels to be recorded, 20 were strain measurement.

Some structural columns were instrumented with strain gauges on opposite sides of the column. These opposing gauges were wired into a single bridge, which permitted the measurement of bending with a single output. This method maximized the information gained from a smaller number of recorded channels. It was necessary to locate the gauges in areas that would not be subjected to local yielding. All strain

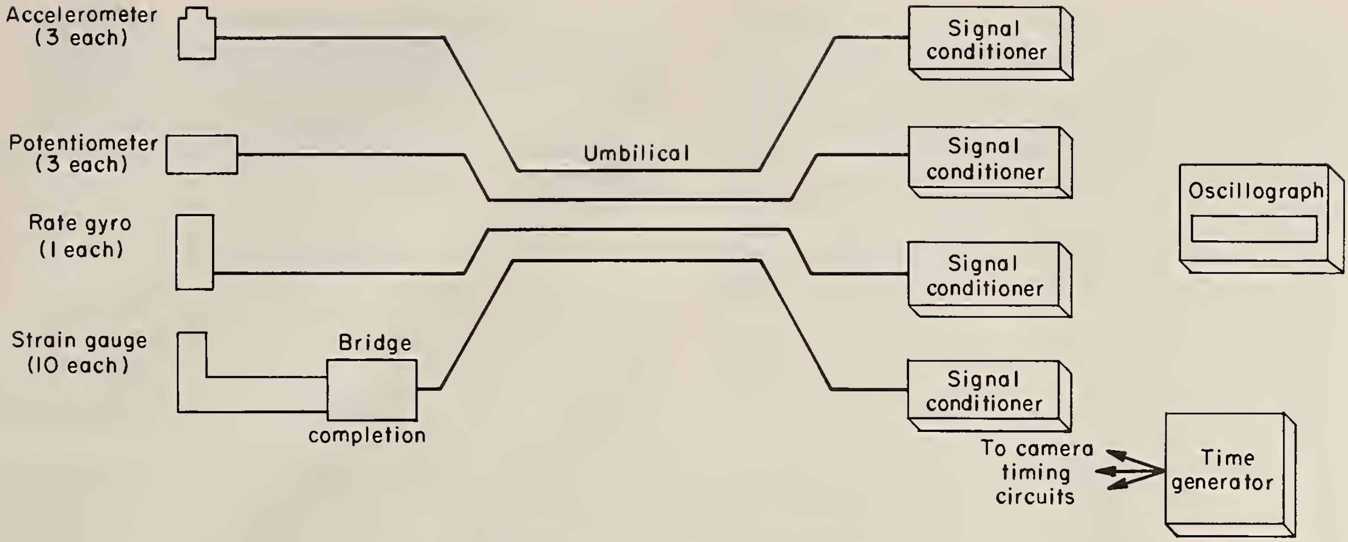

Figure 3. — Instrumentation schematic.

gauges were wired with a three-wire hookup, and precision bridge-completion resistors were located on the machine near the strain gauge locations. All gauges were 1-in uniaxial gauges.

Deflection

Six channels were utilized to measure deflection of particular locations on the ROPS. The measurements were made with resistive, linear position transducers with a 20-in stroke. The diagonal spans across the ROPS columns were measured in three planes to resolve the relative motions of the structure.

Acceleration

Six channels of acceleration were measured and recorded. The accelerometers were mounted in a triaxial configuration at two locations on the machine. The accelerometers had a range of ±20 g.

Roll Rate

This measurement was a direct result of the manufacturer's survey information. One manufacturer was recording roll rates routinely on all roll tests to provide correlation from one test to the next. Roll rate is used to determine differences in roll test results. Acceleration measurements can also be used in this manner, but roll rate is a more positive comparison method. Two channels were utilized to record roll rate.

Time

Each oscillograph recorder was equipped with an internal time base that was recorded on each individual record. In addition, a time base generator for the cameras was recorded on the film edge, as well as on the oscillographic records. Flashbulbs in the view of all cameras were activated at the beginning of the roll to correlate a starting time.

TESTS WITH 52,000-lb-GVW, WHEELED, FRONT-END LOADER

ROLL TEST 1

The rollover test of a 52,000-lb-GVW, wheeled, front-end loader equipped with a four-post ROPS that had a 106-pct-GVW (55,120-lb) side load capability successfully met the objectives of the roll test. The combination of the tilt table and the relative location of the launch pad and the roll hill provided an excellent method for roll initiation, which resulted in a relatively high side load on initial impact. The ROPS was subjected to three impacts during the test. Two impacts occurred on the roll hill and one impact occurred on the bench at the base of the hill.

Table 2 summarizes the data obtained from the test. The data indicate that the structure was subjected to successively greater loads with each impact. The strain gauge data indicate vertical loadings of approximately 200 to 400 pct GVW, 300 to 500 pct GVW, and 700 pct GVW for the first, second, and bench impacts, respectively. The vertical loading estimated from the soil resistance and accelerometers measurements indicates even higher vertical loads.

The strain gauge data indicate a side load of 58,000 and 60,800 lb for the first and the second impacts. The side load could not be determined for the bench impact. The ROPS had a side load capability of approximately 55,100 lb, as determined by a static test conducted by the manufacturers of a similar unit. The measured side loads were thus higher than the estimated capacity of the ROPS. The discrepancies are probably caused by measurement inaccuracies during the static and roll tests, and differences in the cross-sectional geometry and yield strengths of the two ROPS. The side loads for the first two impacts determined from the side deflections of the ROPS were approximately 50,800 and 52,800 lb.

Estimated vertical and side loads, and estimated side deflection, are also given in table 2. These estimates are based on the data obtained from the test, considerations of the reliability and accuracy of the particular measurements, the

capacity of the mounting bolts, and the observations of the high-speed film.

The evaluation of the data obtained from the roll test indicated that the ROPS experienced vertical and side loads that exceeded the requirements of SAE J1040c. For comparison, SAE J1040c requires the following loads and deflections during static testing for the machine and ROPS that were subjected to the roll test: Side load, 73 pct GVW; side deflection, 13 in; and vertical load, 200 pct GVW.

The static test required by SAE J1040c is to provide protection for a 360° roll on a 30° slope. The ROPS that was subjected to the roll test meets these requirements.

From the data and review of the impacts of the test, major structural damage to the ROPS appears to have occurred as a result of the third (bench) impact. The damage included tensile failure of the two 7/8-in, grade 8 mounting bolts at each rear post of the ROPS; tensile failure of the three 3/4-in, grade 8 mounting bolts at the right-front post (the three 3/4-in bolts of the left-front post were not fractured); buckling of the rear plate (between the two rear posts) in the area where an access hole was cut; fracture of the top front crossmember and top plate; yielding of the complete cross section at the top of all four posts just below the gussets; and weld failure at the right mounting bracket for the instrument panel-ROPS attachment. Figure 4 shows the ROPS configuration before and after the roll test.

Table 2.—First roll test summary

(52,000-lb-GVW, wheeled, front-end loader)

	1st impact	2d impact	Bench impact
MEASURED			
Vertical load, 10^3 lb:			
ROPS strain gauges	115–203	147–237	377
Soil resistance	159	192	3,400
Acceleration	208–260	156–312	520
Side load, 10^3 lb:			
ROPS strain gauges	58	60.8	ND
Soil resistance	40	44.3	ND
ROPS deflection [1]	50.8	52.8	ND
Longitudinal load, 10^3 lb:			
ROPS strain gauges	15.14	9.78	ND
Side deflection,[2] in:			
Potentiometers	[3] 4.43	5.25	[3 4] >24
Physical measurement	3.25	3.80	>24
Roll rate,[5] °/s:			
Accelerometers	75	90	160
Gyroscopes	106	120	325
Ground resistance, psi:			
Penetrometer	142	157	3,000
Max ROPS penetration, in:			
Field measurement	24	19	ND
ESTIMATED			
Vertical load:			
Force ....10^3 lb....	160	210	>377
Force ....pct GVW....	310	400	730
Side load:			
Force ....10^3 lb....	51	53	>55
Force ....pct GVW....	98	102	106
Side deflection ....in....	4.4	5.3	24

ND Not determined.
[1] Static test curve.
[2] Plastic 3d impact method.
[3] Maximum.
[4] Estimated, sensors reached maximum at 12 pct of full scale.
[5] Measured using average strain gauges and instantaneous pictures.

The operators platform to which the ROPS was mounted was damaged only in the area of the mounting bolt holes. Other machine damage was minimal. However, the left-rear tire rim seal was damaged and loss of pressure occurred.

As indicated by the ground resistance and ROPS penetration data in table 1, the roll hill was very "soft" and the bench was extremely compacted.

ROLL TEST 2

The second rollover test of the 52,000-lb, wheeled, front-end loader and a four-post ROPS with a 120-pct-GVW side load capability successfully met the objectives of the roll test. Similar to the first test, the ROPS was subjected to two impacts during the test on the roll hill. The ROPS mounts did not fail during the test.

Table 3 summarizes the data obtained from the second test. The data indicate that the ROPS was subjected to successively greater loads with each impact. The strain gauge data indicated vertical loadings of approximately 170 and 180 pct GVW for the first and second impact, respectively. The vertical loadings estimated from the accelerometer measurements for each impact indicated vertical loads of 1.4 and 4.2 pct GVW, respectively. Table 3 also contains estimated vertical and side loads, and estimated side deflection, based on data collected during the test.

The strain gauge data indicated a side load of 57,700 lb (110 pct GVW) and 62,800 lb (120 pct GVW) for the first and second impacts. The measured side load during the second impact is higher than the estimated capacity of the ROPS. The structure had a side capability of approximately 120 pct GVW

Table 3.—Second roll test summary

(52,000-lb-GVW, wheeled, front-end loader)

	1st impact	2d impact
MEASURED		
Vertical load, 10^3 lb:		
ROPS strain gauges	87.45	94
Accelerometer	72.8	218
Side load, 10^3 lb:		
ROPS strain gauges	57.7	62.8
ROPS deflection	49	ND
Longitudinal load, 10^3 lb:		
ROPS strain gauges	10.55	3
Roll rate, °/s:		
Before impact:		
Gyroscope	102	([1])
Film	113	318
After impact:		
Gyroscope	75	([1])
Film	87	273
Max ROPS penetration, in:		
Field measurement	18	16
ESTIMATED		
Vertical load:		
Force ....10^3 lb....	87.5	94
Force ....pct GVW....	170	180
Side load:		
Force ....10^3 lb....	57.7	62.8
Force ....pct GVW....	110	120
Side deflection ....in....	>4.5	>13

ND Not determined.
[1] Off scale.

as determined by the static test results of a similar unit, corrected by the material strength allowables established after the roll test. This estimate is also based on the fact that the mounting bolts that failed during the static test were replaced with standard production bolts.

The side load for the first impact as determined from the side deflections of the structure was approximately 49,000 lb. This side force was determined from the deflections measured from the high-speed film. The displacement transducers appeared to give inaccurate deflection measurements (approximately 1 in) because some of the linear potentiometer mounting studs were bent. Therefore, the side forces determined from the deflections are not as reliable as those determined from the strain gauge data.

Changing the roll hill penetration resistance from 150 psi for the first test to 1,800 psi for the second test appears to have affected the dynamics of the roll more than any other factor. It was expected that an increase in penetration resistance of the hill would have caused an increase in the vertical loads on the structure. In fact, however the vertical loads on the harder hill were only about one-half the loads that resulted from rolling the machine on the softer hill. The harder hill allowed the machine to gain angular momentum from the first to second impact thereby reducing the vertical loads. The angular momentum entering the second impact was 2.8 times greater on the harder hill than on the softer hill. This was observed by comparing the high-speed film of the second impact for both tests.

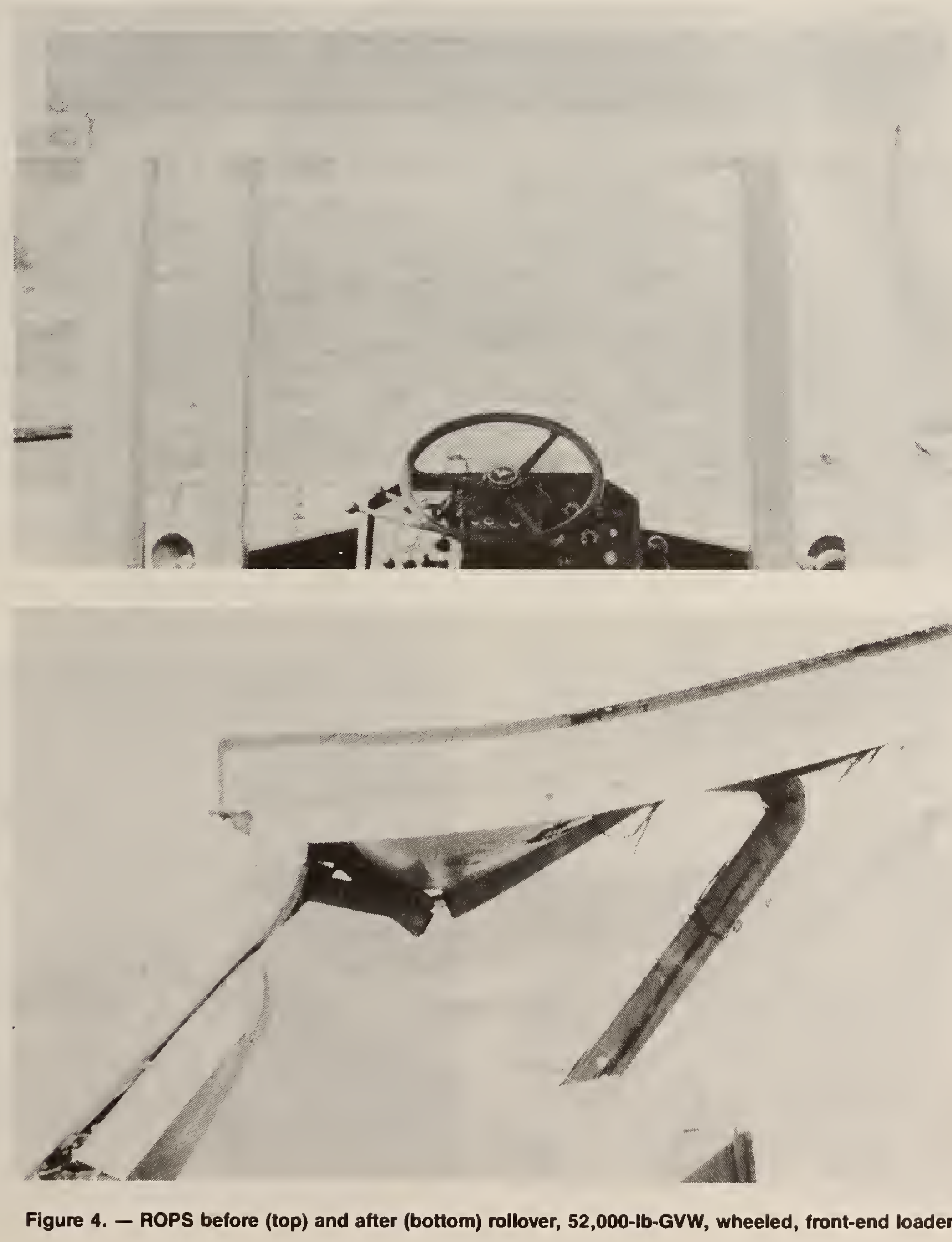

Figure 4. — ROPS before (top) and after (bottom) rollover, 52,000-lb-GVW, wheeled, front-end loader.

TESTS WITH 390,000-lb-GVW, WHEELED, FRONT-END LOADER

ROLL TEST 3

The first rollover test of the 390,000-lb-GVW, wheeled, front-end loader with a four-post ROPS met the objectives of the test. The side of the machine contacted the roll hill before the ROPS did, which introduced maximum side loading of the structure. The machine rolled two complete revolutions on the roll hill, subjecting the structure to two impacts, and landed upright at the base of the hill. The data needed to analyze the behavior of the ROPS during the roll test were obtained and a summary of the results is given in table 4.

The results indicate that the first impact imposed approximately a 110-pct-GVW side load (448,000 lb) on the ROPS. This is supported by the strain gauge data and also from the measured deflections. The strain gauge data show an equal distribution of the side load on the front and rear of the structure. A load-deflection curve generated from the computer analysis provided a means to correlate side deflection to applied load.

The strain gauge data also show approximately a 190-pct-GVW vertical load (727,000 lb) for the first impact. The accelerometer data show a 4.9-g (1,911,000-lb) loading. This loading is predictably higher because only part of the machine weight contributed to the vertical loading. In this test the reduced vertical loading is mainly due to the top corner of the machine's frame absorbing part of the vertical force because it impacted the roll hill first and was still in contact with the ground when the ROPS hit.

The longitudinal load during the first impact, as determined from the strain gauge data, indicates about a 30-pct-GVW rear load (130,000 lb), consistent with previous tests.

Table 4.—Third roll test summary

(390,000-lb-GVW, wheeled, front-end loader)

	1st impact	2d impact
MEASURED		
Vertical load, 10^3 lb:		
ROPS strain gauges	727	1,014
Accelerometer	1,911	1,755
Side load, 10^3 lb:		
ROPS strain gauges	448	398
ROPS deflection [1]	429	526
Longitudinal load, 10^3 lb:		
ROPS strain gauges	58	390
ROPS mount failure	ND	367
Roll rate, °/s:		
Before impact: Gyroscope	108	204
After impact: Gyroscope	42	73
Max ROPS penetration, in:		
Field measurement	47	70
ESTIMATED		
Vertical load:		
Force .... 10^3 lb	727	1,028
Force .... pct GVW	186	264
Side load:		
Force .... 10^3 lb	448	390
Force .... pct GVW	115	100
Longitudinal load:		
Force .... 10^3 lb	130	390
Force .... pct GVW	30	100

[1] Computer load deflection curve.

Between the first and second impacts the machine developed an angle of about 45° to the vertical centerline of the hill, measured from the impact impressions on the hill. Thus, when the ROPS impacted the hill it was subjected to equal longitudinal and side loads, which is confirmed by the strain gauge data. The longitudinal and vertical loads were at approximately 100-pct-GVW (390,000 lb). The vertical load was determined to be about 260-pct-GVW (1,028,000 lb) from the strain gauges.

The ROPS was designed for the following loads: Side load, 175 pct GVW; and longitudinal load, at one-third point on top 99 pct GVW crossmember.

Although the ROPS survived the 360° roll, it did not survive the 720° rollover (fig. 5). The large longitudinal load and the manner in which it was applied to the ROPS during the second impact caused failure of the mount at the left rear post (fig. 6). The loads determined for the second impact are just prior to the failure of this mount. Despite the mount failure, the data indicate the structure was loaded to approximately the designed longitudinal load level. However, the mount failure prevented the ROPS from absorbing most of the energy associated with plastic deformation.

Analysis of the test data and film showed that the longitudinal load for the second impact of this test was not applied in a manner consistent with current design philosophy. It appears that for large machines the ROPS penetration into the soil is extremely deep (70 in for the second impact). The deep penetration in combination with the machine developing a 45° angle to the vertical centerline of the hill in effect caused the ROPS to "plow" through the soil.

The plowing action effectively distributed the longitudinal load along the post and to a lesser extent across the top crossmember of the ROPS. This effectively lowers the height of the resultant load, with the following results:

For loads applied at the top of the ROPS from the rear, the reactions at the bottom of the posts are evenly distributed front and rear.

When the applied load is effectively shifted to one side, the reactions at the bottom of the posts increase on the side that is loaded.

When the applied load is effectively lowered, the reactions at the bottom of the post are redistributed with the horizontal reactions on the rear posts increasing and the horizontal reactions of the front post reduced.

The change in the load application point from the top crossmember (at the one-third or one-quarter point) to along the post does not change the overall capability of the structure.

A failure analysis showed that the mounts had the capability of supporting a 150-pct-GVW longitudinal load if the load was applied at the top crossmember of the structure. That is, the mounts were 50 pct stronger than the ROPS. However, for a load distributed on the ROPS post, simulating the test, the capability of mounts (94 pct GVW) is slightly less than that of the ROPS. The strain gauge data indicated that the ROPS was subjected to a 100-pct-GVW longitudinal load before the mount failed.

The results of this rollover test confirmed a design philosophy that ROPS mounts must have the structural capability to allow large plastic deformations in order to absorb the impact energy. However, the location of the longitudinal impact load on the ROPS did not occur as anticipated. The deep penetration into the soil, in combination with the machine developing a 45° angle, caused the longitudinal load to be distributed

Figure 5. — First 390,000-lb-GVW, wheeled, front-end loader after rollover.

Figure 6. — Mount failure, 390,000-lb-GVW, wheeled, front-end loader.

along the post instead of along the top crossmember. This effectively concentrated more of the load on one post instead of a more even distribution on all four posts.

ROLL TEST 4

The second roll test of the 390,000-lb-GVW, wheeled, front-end loader met the objectives of the test. Data were obtained to determine the magnitudes and direction of loading on the ROPS (table 5). The machine rolled two complete revolutions and subjected the ROPS to two impacts on the roll hill.

The machine dynamics were very similar to those of the previous test of the same machine. The major difference was the fact that the longitudinal centerline of the machine only changed about 19° before the second impact as opposed to 45° in the previous test. This meant that there was a smaller longitudinal load on the ROPS during the second impact. Figure 7 shows the ROPS before and after the test.

The strain gauges provided impact load data for the vertical, side, and longitudinal directions. The displacement potentiometers provided data to determine loads for only the

Table 5.—Fourth roll test summary

(390,000-lb-GVW, wheeled, front-end loader)

	1st impact	2d impact
MEASURED		
Vertical load, 10^3 lb:		
ROPS strain gauges	811	1,015
Accelerometer	801	1,158
Side load, 10^3 lb:		
ROPS strain gauges	604	616
ROPS deflection [1]	508	576
Longitudinal load, 10^3 lb:		
ROPS strain gauges	50	189
Roll rate, °/s:		
Before impact: Gyroscope	118	218
After impact: Gyroscope	40	67
Max ROPS penetration, in:		
Field measurement	57	84
ESTIMATED		
Vertical load:		
Force ..... 10^3 lb....	811	1,092
Force ..... pct GVW....	210	280
Side load:		
Force ..... 10^3 lb....	604	616
Force ..... pct GVW....	155	158
Longitudinal load:		
Force ..... 10^3 lb....	50	189
Force ..... pct GVW....	13	48
Av side deflection ..... in....	1.5	3.0

[1] Static test curve.

side direction and accelerometer data provided loads for the vertical direction. The strain gauges provided a means to determine the loads by a standard method of data reduction. However, the reduction of the accelerometer data and the displacement potentiometer data required consideration of the machine dynamics in order to obtain meaningful results.

In using the accelerometer data to calculate a vertical load, an effective mass of the machine acting on the ROPS during the impacts had to be assumed. Correlating the vertical loads and accelerometer data from previous tests indicates that for this test approximately 40 pct of the machine weight contributed to the vertical load on the ROPS. Therefore, 40 pct of the machine mass times the measured vertical acceleration of the machine was used to calculate the vertical load on the ROPS. This correlated closely with the strain gauge data.

The deflection data are difficult to use in determining the applied side load. The location of the side load during a roll test and static test will invariably be different. Thus, the deflections measured during a roll test cannot be directly used to correlate loads using the static test curve. However, an attempt was made to estimate the effects of the different load locations. In addition, the deflections were adjusted because of the geometry effects caused by potentiometer locations.

STATIC TEST OF FOUR-POST ROPS

The static testing of the four-post ROPS for the 390,000-lb-GVW, wheeled, front-end loader met the objectives of the

Figure 7. — Second 390,000-lb-GVW, wheeled, front-end loader before (top) and after (bottom) rollover.

test. Load deflection curves were obtained for the side, vertical, and longitudinal directions. The energy absorbed by the ROPS was determined for the side and the longitudinal load tests. The loading sequence was as follows:

Side-load.—On the side opposite the roll test loads to minimize the effects of possible frame damage.

Vertical load.

Longitudinal load.—Rear load at the one-quarter point on the same side of the ROPS as the side load.

During the side load test the ROPS was subjected to a load of 685,100 lb. The ROPS deflected a total of 8.5 in and absorbed 4,179,000 in•lb of energy. The load deflection curve for the side load test is shown in figure 8. The frame at the ROPS-machine innerface deflected a maximum of 0.85 in and had a permanent deflection of 0.26 in. Table 6 shows the side load energy for each displacement.

A load of 1,567,000 lb was applied to the ROPS during the vertical load test. As shown in figure 9, this resulted in a vertical deflection of 0.24 in.

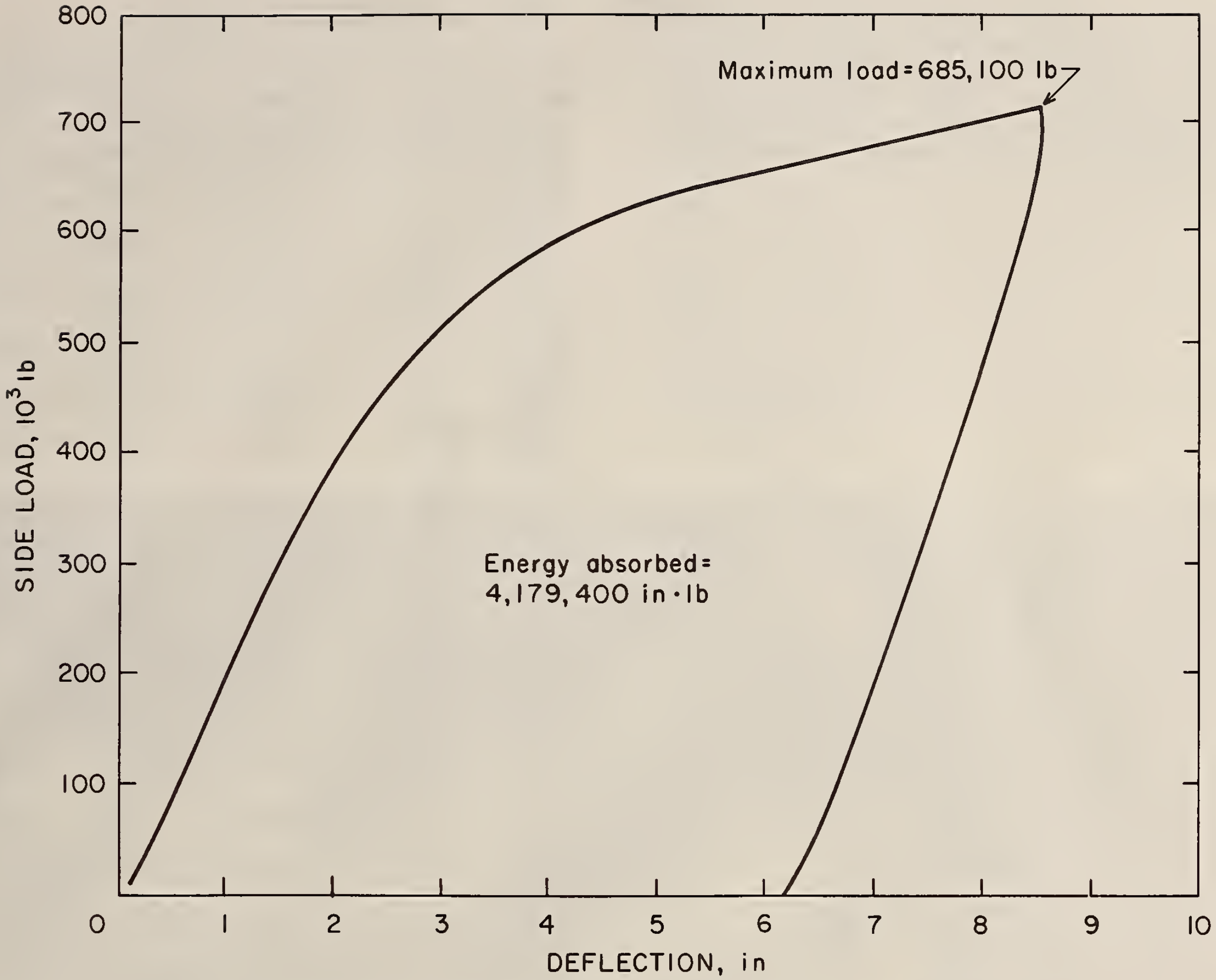

Figure 8. — Static test side load deflection curve, 390,000-lb-GVW, wheeled, front-end loader.

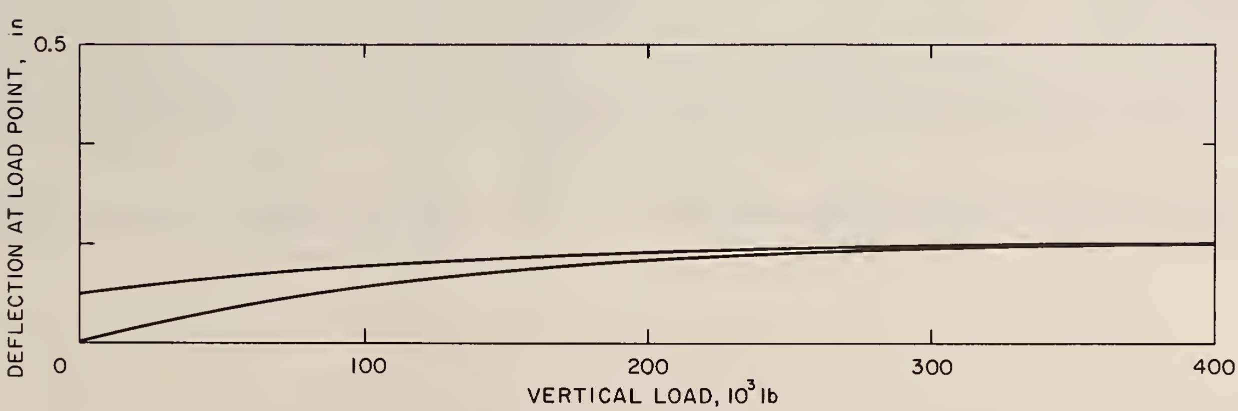

Figure 9. — Static test vertical deflection curve, 390,000-lb-GVW, wheeled, front-end loader.

During the longitudinal load test a load of 659,000 lb was applied to the ROPS. The ROPS deflected >3 in and it absorbed 1,176,000 in•lb of energy. The load deflection curve for the rear load is shown in figure 10. The rear load energy is shown in table 6. The capacity of the hydraulic system was reached during the test, which prevented any higher loads from being reached.

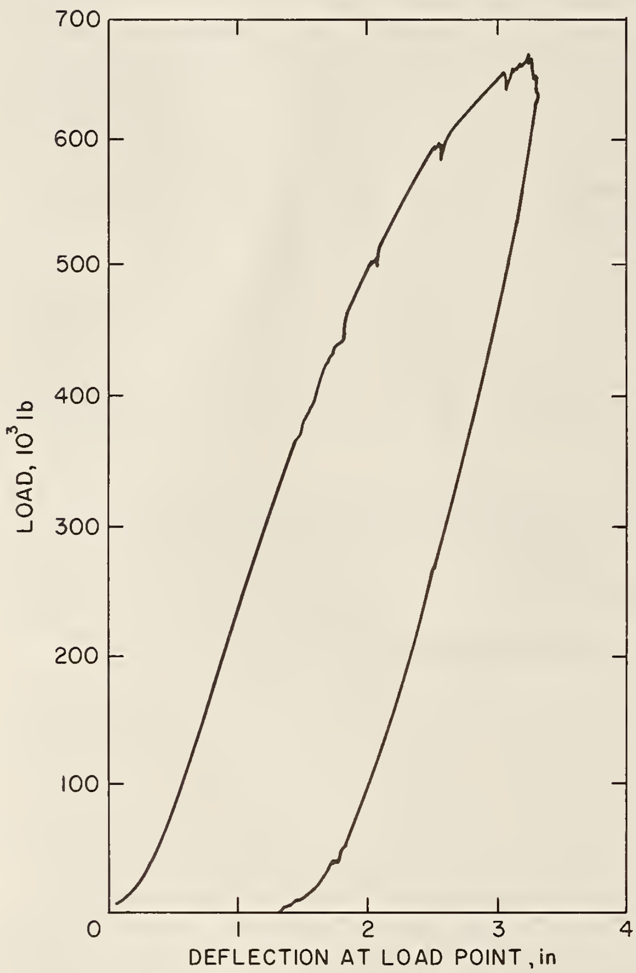

Figure 10. — Static test longitudinal load deflection curve, 390,000-lb-GVW, wheeled, front-end loader.

Table 6.—ROPS static test side and longitudinal load energy

Load, 10^3 lb	Displacement, in	Av load, 10^3 lb	Energy, 10^3 in-lb: Incremental	Energy, 10^3 in-lb: Cumulative
SIDE LOAD, 6.03-in FINAL DISPLACEMENT				
0	0	} 43.19	21.60	21.60
86.37	.50			
86.37	.50	} 141.88	71.65	93.24
197.40	1.01			
197.40	1.01	} 345.58	694.27	787.51
493.77	3.01			
493.77	3.01	} 513.74	252.76	1,040.27
533.70	3.51			
533.70	3.51	} 551.60	273.59	1,313.87
569.50	4.00			
569.50	4.00	} 581.98	293.32	1,607.19
594.47	4.51			
594.47	4.51	} 599.84	299.32	1,906.51
605.20	5.01			
605.20	5.01	} 612.30	307.99	2,214.49
619.40	5.51			
619.40	5.51	} 626.18	311.21	2,525.70
632.97	6.01			
632.97	6.01	} 638.44	321.14	2,846.84
643.90	6.51			
643.90	6.51	} 648.04	305.87	3,152.71
652.17	6.98			
652.17	6.98	} 658.48	346.36	3,499.07
664.80	7.51			
664.80	7.51	} 672.54	334.93	3,834.00
680.27	8.00			
680.27	8.00	} 682.68	345.44	4,179.44
685.10	8.51			
685.10	8.51	} 342.55	847.81	3,331.63
0	6.04			
LONGITUDINAL LOAD, 1.18-in FINAL DISPLACEMENT				
0	0	} 39.14	18.55	18.55
78.27	0.47			
78.27	0.47	} 156.07	76.47	95.02
233.87	.96			
233.87	.96	} 308.78	159.95	254.97
383.70	1.48			
383.70	1.48	} 441.26	217.98	472.96
498.83	1.98			
498.83	1.98	} 544.23	269.94	742.90
589.63	2.47			
589.63	2.47	} 621.33	311.29	1,054.18
653.03	2.97			
653.03	2.97	} 656.20	122.05	1,176.23
659.37	3.16			
659.37	3.16	} 329.68	−651.12	525.12
0	1.18			

TEST WITH 286,000-lb-GVW, WHEELED, FRONT-END LOADER

The rollover test of the 286,000-lb-GVW, wheeled, front-end loader (test 5) met the objectives of the test. Data were obtained to determine the magnitudes and directions of loading on the ROPS. The machine rolled two complete revolutions and subjected the ROPS to two impacts on the roll hill. An additional ROPS impact occurred when the machine came to rest on its side on the road at the bottom of the hill as shown in figure 11. This impact was softened because the road had been ripped prior to the roll. In addition, the tires had blown out on the machine, which caused a large decrease in its roll rate prior to impact. The only damage to the machine frame or ROPS mounts that was observed was the blown tires and bent rims on the left side of the machine.

During the roll the longitudinal centerline of the machine remained perpendicular to the vertical centerline of the hill. This resulted in low longitudinal loading on the ROPS. A summary of the ROPS loading is given in table 7.

The displacement potentiometers provided data to determine loads for the side and longitudinal direction. Accelerometer data were used to obtain loads for the vertical direction. The strain gauges provided impact load data for the vertical, side, and longitudinal directions. The strain gauges provided a means to determine the loads at a standard method of data reduction. The reduction of the accelerometer data required consideration of the machine dynamics in order to obtain meaningful results. The potentiometer data were corrected for the geometery effects in their readings caused by the location of the potentiometers.

The ROPS side loading during the first two impacts on the roll hill was much lower than expected. In order to obtain

Figure 11. — Roll test 5, 286,000-lb-GVW, wheeled, front-end loader. Note that an anthropometric dummy was strapped in the cab to evaluate the Bureau's Vest Restraint System.

an understanding of the low side loads a quantative analysis was conducted of the following factors that influence the dynamics of a roll:

Machine mass.
Radius of gyration.
Location of the center of gravity.
Distance of the ROPS from the center of gravity.
Change in roll rates during impacts.
Translational velocity of the machine.
Penetration resistance of the roll hill.

A review of these factors does not indicate any gross difference with the previous roll tests that would explain the low loads. It appears that the major difference in the factors is the location of the center of gravity. The 286,000-lb-GVW machine appears to have a much lower center of gravity. The machines previously tested rolled off the tilt table at approximately a 50° angle. However, this machine rolled off at approximately a 65° angle and entered the first impact at a slightly higher rotational velocity. From an analysis of the films it appears that the machine was in a more vertical position when the first impact occurred.

An examination of the overall configuration of the ROPS and the ROPS-machine configuration does suggest some differences which may account for the low loads. For example, the ROPS on the 286,000-lb-GVW machine differs from the 390,000-lb machine in overall dimensions such as follows:

Smaller plan view area because of slanted posts and the location of the operator within the ROPS.

Smaller side profile area (no crossmember and shorter length) because of the plate construction.

Also, overall machine parameters, such as ROPS-machine width ratio and ROPS-machine height, differed between the machines tested.

A qualitative comparison of these factors for various machines presently available may indicate that there are significant differences between the 286,000-lb-GVW machine with the present ROPS configuration and the other machines.

A review of the motion picture coverage of the test indicates that the generally smaller profile of this ROPS, especially the thin (1 in) top plate, may have resulted in the ROPS slicing through the soil and offering very little lateral area to develop a side force. On the first impact the ROPS posts left a clear imprint in the hill indicating the side forces developed lower on the top plate. This is clearly seen for the first 0.05 s where little side load was experienced. For this time period only the top plate was in contact with the ground.

Table 7.—Fifth roll test summary

(286,000-lb-GVW, wheeled, front-end loader)

	1st impact	2d impact	Bench impact
MEASURED			
Vertical load, 10^3 lb:			
ROPS strain gauges	562	559	NAp
Accelerometer	313	627	NAp
Side load, 10^3 lb:			
ROPS strain gauges	226	192	NAp
ROPS deflection [1]	180	180	NAp
Longitudinal load, 10^3 lb:			
ROPS strain gauges	65	90	NAp
Roll rate, °/s:			
Before impact: Gyroscope	125	241	NAp
After impact: Gyroscope	63	165	NAp
Max ROPS penetration, in:			
Field measurement	22	38	NAp
ESTIMATED [2]			
Vertical load:			
Force ... 10^3 lb	562	559	<7.3
Force ... pct GVW	≈197	196	<3
Side load:			
Force ... 10^3 lb	226	192	<52
Force ... pct GVW	79	67	<18
Longitudinal load:			
Force ... 10^3 lb	65	90	<12.8
Force ... pct GVW	23	32	<8
Side deflection ... in	1	1	NAp
Longitudinal deflection ... in	0.6	ND	NAp

NAp Not applicable. ND Not determined.
[1] Computer load deflection curve.
[2] Data are rounded.

CONCLUSIONS

It should be recognized that ROPS performance criteria do not guarantee protection of the operator in all cases. From the testing conducted during this program for example, it appears that when machines lose contact with the hill and become airborne, even larger forces and energy demands are imposed upon the ROPS structure. Thus, care must be taken to ensure that test conditions and/or field usage do not invalidate performance criteria.

A longitudinal force and energy capability has never been specifically addressed in ROPS performance criteria. Although the tests conducted for this program were designed to provide maximum side loading and only minimum longitudinal loading, longitudinal loads were recorded for each test. With the exception of one ROPS impact, the longitudinal loads were relatively low in comparison to the side load. The second ROPS impact of test 3 did cause the ROPS to fail because of the location of the longitudinal load resulting from the orientation of the machine relative to the downward slope of the roll hill.

Larger longitudinal loads than were observed during the roll tests are possible in an actual field rollover. For example, if prior to a rollover a machine was being operated in one of the following situations larger longitudinal loads could result:

1. In a position pointing slightly uphill or downhill.
2. If an operator abruptly changed steering direction.
3. If the machine was traveling at an appreciable forward or aft velocity.
4. If a small machine was suddenly accelerated causing rear upset because of rotation about the powered axle.
5. In an accident during loading or unloading onto a lowboy truck.

In any case, it would be anticipated that the principal roll axis would rapidly revert to the longitudinal axis of the machine, thus subjecting the ROPS to greater side loads than longitudinal loads. Therefore, the longitudinal loading and energy criteria should be equal to or less than the side load capability. It is believed that a longitudinal force that is 80 pct

of the side criteria would be adequate for 540° rolls (two top and two side loads).

The other major performance factor to be considered is the vertical load. SAE J1040c criteria currently require a minimum 200-pct-GVW vertical load for all machine types and masses. This requirement is usually met or exceeded as a result of the high vertical stiffness and load carrying capability required to meet load and energy criteria in the critical side direction. A ROPS generally has a vertical capability in excess of 200 pct GVW and maybe as high as 1,000 pct GVW even though the roll test data indicate actual maximum vertical loads of only 200 to 400 pct GVW.

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